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Research article

Numerical analysis of the impact of winding angles on the mechanical performance of filament wound type 4 composite pressure vessels for compressed hydrogen gas storage

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ABSTRACT

Transportation relies heavily on petroleum products, forcing the adoption of alternative energy sources like hydrogen. Hydrogen is considered the cleanest fuel for the twenty-first century due to its water-based combustion and no CO_2 emissions. However, challenges persist in production, utilization, and storage; employing composite material-based high-pressure storage vessels is increasing in the hydrogen storage sector. The paper analyzes the impact of the winding angles on the mechanical performance of the filament wound Type 4 composite pressure vessels (CPVs) for compressed hydrogen gas storage at 70 MPa. This work examines the individual winding angles and combined angles winding patterns to promote the efficiency of Type 4 CPVs by achieving maximum burst pressure, ensuring safe burst mode, and reducing CPV weight by applying maximum principal stress theory with the aid of the Ansys ACP Prep/Post and static modules. The weight and burst pressure of CPVs are significantly influenced by fiber orientation; a combination of positive and negative helical winding angles promotes higher burst pressure at a lower weight. A hoop angle and intermediate helical angles can be combined to create high-efficiency CPVs that provide mechanical performance comparable to that of a combination of high and low helical angles. Finally, a one-factor-at-a-time (OAT) sensitivity analysis was performed to determine how the winding angle and the thicknesses of layers affect the CPVs' performance. It was found that the performance of the CPVs is significantly influenced by the thicknesses of the wound layers.

1. Introduction

The main approach to mitigating global warming and climate change is to reduce CO_2 emissions from burning fossil fuels in different transportation systems. Furthermore, the abundance of oil, coal, and natural gas is limited, necessitating alternative solutions before the demand can no longer be met [1–7]. In this situation, hydrogen has a lot of potential as an energy carrier. A hydrogen economy is justified by the fact that hydrogen fuel is clean, environmentally friendly, and produces very little or no greenhouse gases [1–9]. Fig. 1 shows some applications of hydrogen as a fuel for various kinds of sustainable green transportation systems, including

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heavy-duty vehicles (Fig. 1 a) and aircraft (Fig. 1 b) [10]. In the current vehicle fuel market, hydrogen is still in the early stages of development [1,5,6,9]. The key elements of a hydrogen economy are hydrogen production, storage, distribution, and utilization [2,3,6,7]. The United States Department of Energy (DOE) lists weight, volume, durability, efficiency, cost, refueling time, and life cycle as the most essential technological and financial obstacles for hydrogen-powered transportation [1]. The development of a practical, safe, and effective approach to the storage of hydrogen remains the most difficult task, even though these aspects have been the focus of intensive research for decades.

Hydrogen can be produced using different methods, including the electrolysis of water, the reforming of methane and methanol, or the evolution of hydrogen in biomass reactors [6,7,11]. It can then be directly combined with oxygen to create electricity in fuel cells or used to generate mechanical energy in combustion engines [5,6,11,12]. Several studies have investigated the various ways of storing hydrogen in its different states [11,13]. However, to make hydrogen energy a reality, a broader understanding of technology is required.

Hydrogen can be stored in gaseous, liquid, cryo-compressed, or solid-state forms, each with advantages and drawbacks of its own [3,6,13]. For gaseous form, high-pressure gas containers (up to 70 MPa) are used for both stationary purposes and onboard applications when high capacity is demanded [1,3,5,8,13]. In the past few decades, a variety of pressure vessels have been developed to store hydrogen, with the main requirement of weight savings at high-pressure ratings [8]. There are four types of pressure vessels, as presented in Fig. 2. *Type 1* is a seamless or welded metallic pressure vessel made of steel or aluminum (Fig. 2 a). This type is utilized for less than about 30 MPa pressure. In addition to its heavy weight, this type is susceptible to hydrogen embrittlement. *Type 2* pressure vessels have a thick metallic liner that is covered in a fiber composite on the straight cylindrical side without a dome (Fig. 2 b). This type has no pressure limit and has about 35 % less weight. *Type 3* builds up a metallic liner and is wholly wound in a fiber composite, including the dome (Fig. 2 c) [3,5–9,14]. *Type 4* comprises a polymeric liner, usually made from nylon or high-density polyethylene, fully wound with a carbon/glass fiber-resin system (Fig. 2 d) [3,5,14]. Both Type 3 and Type 4 composite pressure vessels (CPVs) can withstand up to 70 MPa working pressure. Type 4 CPVs save 75 % of their weight compared to metallic vessels, which makes them an excellent choice for storing compressed hydrogen gas for use in transportation systems because they will consume less fuel due to their lower weight, thereby raising the fuel efficiency [1,2,5,6,8,12,15]. CPVs also have higher burst pressure which makes them safer compared to Type 1. All CPVs employ a liner to act as an obstacle against gas leakage and provide strength and defense against burst failure [1,6,9,12,14]. In Type 4 CPVs, the composite layers carry the full load as the polymeric liner does not share the load, in contrast to Type 3 CPVs with metallic liners [9,12].

Filament winding technology is the best technique for the fabrication of Type 4 CPVs. Continuous fiber filaments are wrapped around the mandrel after being impregnated with resin [6,15,16]. Various fibers can be used to create filament wound pressure vessels in a polymer matrix [4,7,11,15,16]. Epoxy is a polymer matrix material that is frequently used in composites because of its remarkable strength, hardness, chemical resistance, and compatibility with a wide variety of fiber types [1,12,16,17]. In filament-wound composites, the matrix uniformly distributes the load while the reinforcing material serves as a load-bearing part [7,14,16,17]. The matrix keeps the fibers in place and protects them from outside circumstances [15]. The wound fiber is often formed from materials such as carbon fiber or glass fibers, which provide the vessel with the required rigidity and durability against internal pressure. Carbon fibers are a favorable material for strengthening because they are resistant to corrosion and fatigue [1,2,6,12,16]. The best material for compressed hydrogen gas storage is carbon fiber reinforced plastic (CFRP) composite pressure vessels because they are more durable, have a greater strength-to-weight ratio, and have a longer service life than other types of pressure vessels [4,5,12,14]. Numerous studies have examined how the filament winding process's processing factors affect CPV performance [15,16].

The internal pressure results in high mechanical residual stresses, which makes substantial utilization of carbon fiber necessary. 50 %–70 % of the vessel's total cost is made up of composite materials [3,8]. Research and development are required for this technology to reduce costs and enhance the performance, reliability, and durability of high-pressure CPVs. A key factor in customizing the pressure-bearing characteristics of CPVs is the winding angles [1].

The burst response test assesses a pressure vessel's ability to withstand a sudden increase in pressure above its design pressure. Pressure vessels are intended to contain a specific volume of pressurized gas. The stress response is a crucial safety feature since a pressure vessel collapse can seriously injure or even kill living things. It might also damage the equipment itself, the surrounding environment, and the infrastructure that the device is a part of [7,12]. With the help of its pressure relief device, a pressure vessel's

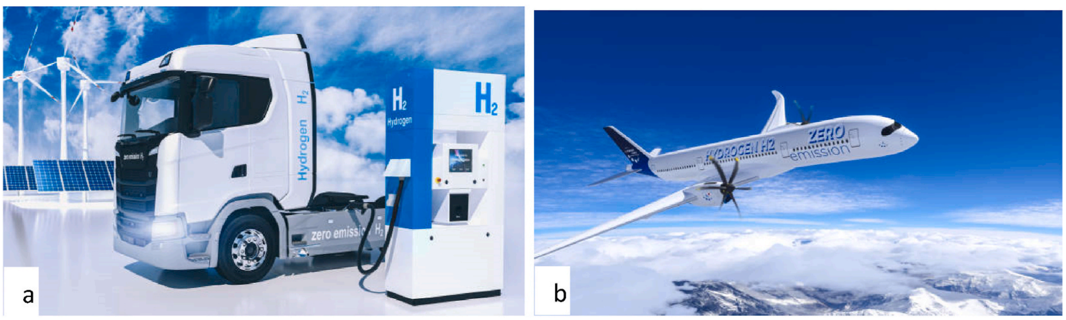

Fig. 1. Applications of hydrogen as a fuel in various kinds of sustainable green transportation systems [10]: (a) heavy-duty vehicles and (b) aircraft.

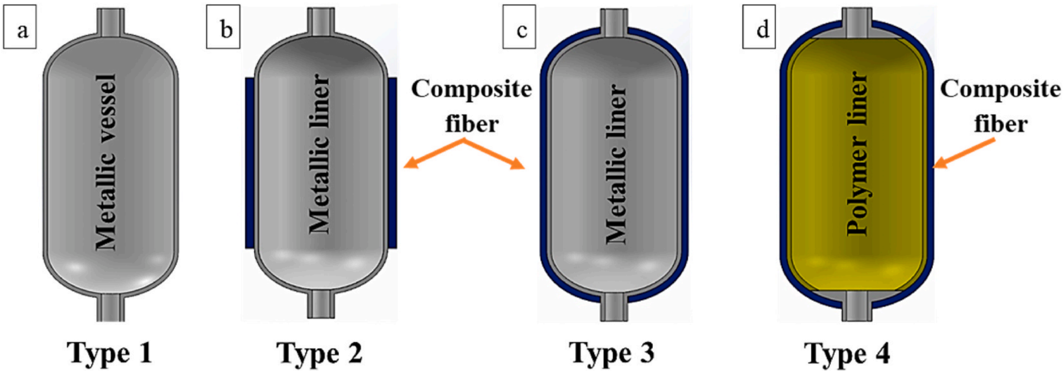

Fig. 2. Types of pressure vessels: (a) Type 1, (b) Type 2, (c) Type 3, and (d) Type 4.

valve is crucial in burst failure. As it maintains a gradual release of the gas instead of an explosion. Therefore, for the pressure vessels, the favored location of failure is the cylindrical portion. To reduce the possibility of boss failure during the burst, the vessel must be designed to fail at the cylindrical portion, which is termed safe burst mode [3,4,9,17].

However, CPVs have unique issues with stress levels and burst response due to the anisotropic structure of the fibers and matrix. Maintaining the reliability and safety of CPVs requires a close assessment of their stress levels. Analytical models, experimental testing, and numerical simulations can all be used to accomplish this. Knowing how various factors affect performance might help with the design and production of CPVs for particular uses [4,12,18]. A CPV's performance is dependent on many factors, including the basic materials used to manufacture the liner and the composite, the geometry of the liner, the winding configuration of the composite layers, the filament winding process parameters and sequence, and the curing aspects [3,4,7,12,17]. Since the composite is wound around the entire vessel in Type 4 CPVs, winding feasibility is a key factor in defining the dome profile. A pressure vessel with a variety of dome heads will have varying mechanical characteristics. Fig. 3(a–e) presents different profiles of the pressure vessel domes. The ellipsoid dome types are the intermediary geometries between an isotensoid dome, which is used to prevent fiber slippage, and a hemispherical head, which is used to support the maximum load [3,17]. For winding composites utilizing filament winding technology without the possibility of fiber slippage, an isotensoidal dome head is thought to be a suitable candidate.

In the case of automotive applications, cylinders with the least weight and high burst pressure help to increase fuel efficiency. According to ISO 15869 standards, the pressure vessel is used to store compressed hydrogen gas at a working pressure of 35 up to 70 MPa [3,5,6,9,14,17]. Research on the impact of the angle winding in CPVs is extremely limited [1,12,14]. FEA is a useful tool for analyzing how complicated structures behave under different loading loads; therefore, the behavior of CPVs in real service conditions can be simulated using finite element analysis (FEA) [7,12,14].

In the current study, tailoring of the winding configuration (angle, orientation, arrangement, and number of layers) of a filament wound Type 4 CPVs used for compressed hydrogen gas storage at an internal pressure of 70 MPa was undertaken using the Ansys ACP Pre/Post and static modules to study the stress distribution of the CPVs and to predict their burst pressures. Moreover, a sensitivity analysis was performed to measure the influence of the winding angle and the thickness of layers on the performance of the Type 4 CPVs.

2. Concept and scope

Fuel cell car suppliers now choose to use compressed hydrogen gas as a fuel [5,6,17,18]. By considerably increasing vehicle efficiency in mobile storage applications, the vessel's weight has a major impact on its performance [17]. The pressure vessel used to store hydrogen must meet two key requirements: high burst pressure and high weight performance. Currently, CPVs are a dependable and well-respected option for compressed hydrogen gas storage on board. High burst pressure is provided by a Type 4 CPV with a

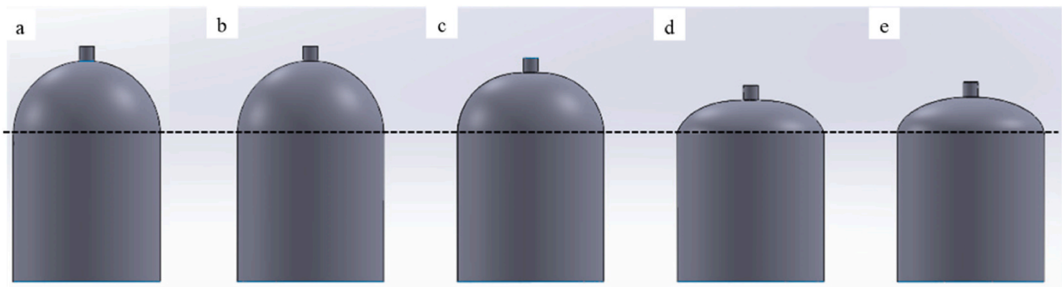

Fig. 3. Different profiles of the pressure vessel domes: (a) hemispherical, (b) ellipsoid (I), (c) ellipsoid (II), (d) ellipsoid (III), and (e) isotensoid.

polymer liner and carbon fiber/epoxy composite while retaining its lightweight [2,9,18]. Many factors influence these features in CPVs, but the vessel's geometry and the composite's winding configurations have a major impact on these characteristics.

The winding angle and the fiber orientation have a significant impact on the mechanical properties of the vessel, resulting in variations in the vessel's strength and stiffness along different axes. The burst behavior is significantly influenced by the distribution of stresses inside the vessel, which emanate from the orientations and characteristics of the wound fibers [1,12]. Fig. 4 shows a schematic illustration of the various winding angle orientations with respect to the vessel's primary axis. Various fiber winding orientations, including helical, hoop, and polar windings, are employed in CPVs, and each offers potential advantages [1,12,16]. The efficiency of the vessel is greatly impacted by these orientations. A polar angle (0°) counteracts axial stress and boosts strength in the axial direction, whereas a hoop angle (90°) resists hoop stress and enhances strength in the hoop direction. Intermediate angles result in stress resistance to both axial and hoop stresses [1,12]. The intended CPV characteristics will determine which fiber winding orientation is selected [12]. A section of the Type 4 CPV with the liner and composite wound layers is displayed in Fig. 5.

Because the composite material used in filament-wound Type 4 CPVs is anisotropic, the stresses are distributed differently in these structures. Hoop stress refers to the stress a material experiences in the radial direction, while axial stress is the stress it experiences in the longitudinal direction. Overweight vessels arise when isotropic materials, such as metals, are utilized in the vessel's construction, as the material is not entirely utilized in the longitudinal direction. However, in a filament-wound composite, the winding angle can be adjusted to optimize the vessel performance in both directions. Since they are the primary factors in determining burst pressure, hoop stress, and axial stress are therefore the two crucial components of pressure vessel analysis that guarantee safety and integrity [12]. The exact values of these stresses depend on various factors, such as the internal pressure, the material properties, the winding angle, and the geometry of the vessel.

Generally, in a material under load, various stresses act at different angles and directions. Principal stresses are normal stresses (tensile or compressive) acting on specific orientations where there are no shear stresses. The highest normal stress encountered at a particular location within the material is referred to as the maximum principal stress. The key terms used to determine the maximum principal stress value are hoop and axial stresses. Estimating the maximum principal stress is vital because it provides valuable insights into the material's response to loads and its potential for failure. One of the failure theories is the maximum principal stress theory, which uses the maximum principal stress to predict the potential of material failure based on its ultimate strength. Analyzing the distribution of the maximum principal stress across the CPV helps identify locations most susceptible to failure, aiding in design optimization.

Tailoring the fiber winding maximizes the weight efficiency of the vessel and minimizes material waste. The current work investigated the influence of the winding angle and stacking arrangement to reduce the number of layers and achieve high and safe burst pressure. Using the maximum principal stress failure theory, the intended pressure vessel's burst pressure is estimated through FEA.

3. Materials and method

3.1. Liner and composite material characteristics

In this work, nylon 6 is used as a liner material in Type 4 CPVs. The mechanical properties of the liner materials are listed in Table 1. The profile and dimensions of the liner are shown in Fig. 6. The outer diameter of the liner is 305 mm, and the length is 1026 mm, with a 5.17 mm thickness. The design of safe pressure vessels with the fewest possible composite layers can result from the ability to prevent fiber slippage and wind the stressed fibers with equal tension. Therefore, a geodesic dome is employed to enhance the winding efficiency of Type 4 CPVs. The geometry is created following the specifications for use as an automobile storage device [12].

As a result of their low density, low elongation at break, and high tensile strength, carbon fibers are used as reinforcement materials in CPVs. Epoxy resins are commonly used as the matrix material in high-performing composites because of their high shear strengths [17]. In the current study, Type 4 CPVs are built using epoxy-carbon unidirectional prepreg composite materials. Table 2 lists the mechanical characteristics of unidirectional prepreg composite materials made of epoxy and carbon fibers.

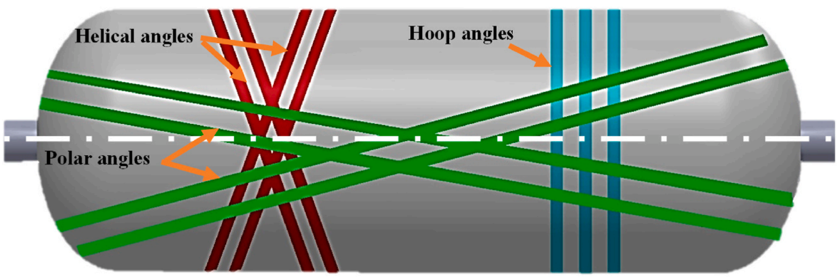

Fig. 4. Schematic representation of the different orientations of the winding angles regarding the major axis of the vessel.

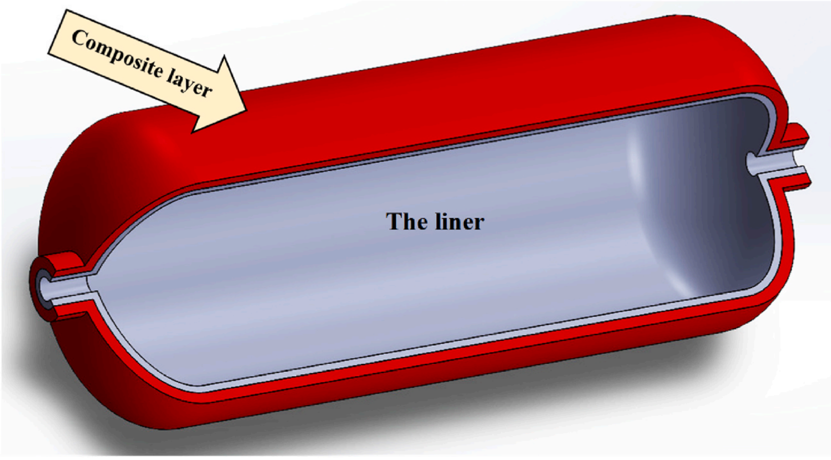

Fig. 5. Section of Type 4 CPV showing the liner and the composite layers.

Table 1
Mechanical properties of Nylon 6 liner.

Density	1140 kg m^{-3}
Coefficient of thermal expansion	0.000147 C^{-1}
Young's modulus	1060 MPa
Poisson's Ratio	0.35
Bulk modulus	1178 MPa
Shear modulus	392.59 MPa
Tensile yield strength	43.1 MPa
Tensile ultimate strength	49.7 MPa
Isotropic thermal conductivity	0.243 J m^{-1}. S. C^{-1}
Specific heat	1500 J. Kg^{-1}. C^{-1}
Isotropic resistivity	1.83*10^{12} Ω m

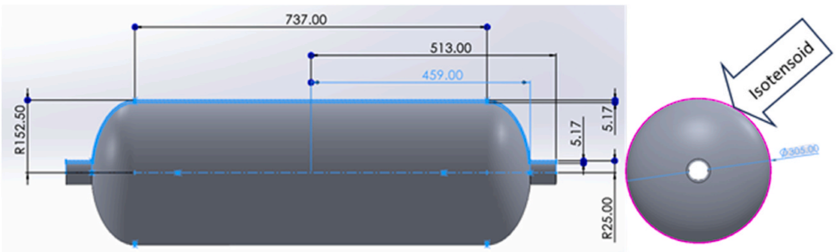

Fig. 6. Profile and dimensions of the liner.

3.2. *Finite element analysis (FEA) and internal pressure*

The main barrier to the use of CPVs is their cost of production and evaluation. Therefore, the FEA approach is used to predict the stress distribution and burst pressure to guarantee that the vessel can sustain the desired operating conditions without failure [12]. Researchers can alter the winding angle in the FEA model to see how the composite structure reacts. This may contribute to improving the CPVs' overall performance [7,9,12–15,18].

A specialized tool in Ansys for the design and analysis of multilayer composite structures is the Ansys Composite PrepPost (ACP) module. With the use of the ACP module, designers can produce parametric design assessments to evaluate the impact of particular parameters on the structure and functionality of CPVs, such as the number of layers, fiber orientation, and fiber arrangement. By applying the proper rosettes to the oriented selection sets, the ACP PrepPost module was used to build modeling plies in certain directions following the predetermined winding patterns. Some winding angles orientation sets on the Ansys ACP Pre/Post module are shown in Fig. 7(a–d). In the current work, the Ansys static module was utilized to study the stress distribution of the Type 4 CPVs and to predict their burst pressures. Building the geometrical model, defining boundary conditions, meshing, interactions, and parameter estimation based on the actual situation are all defined in the analysis of FE models. A three-dimensional vessel was modeled (Fig. 6), and appropriate symmetric constraints were given. Regarding the boundary conditions of the FEM, fixed support was applied to one

Table 2
Mechanical properties of Epoxy-carbon unidirectional prepreg composite materials.

Density	1540 kg m^{-3}
Coefficient of thermal expansion X direction	-4 $*10^{7}$ C^{-1}
Coefficient of thermal expansion Y direction	3 $*10^{5}$ C^{-1}
Coefficient of thermal expansion Z direction	3 $*10^{5}$ C^{-1}
Young's modulus X direction	209 GPa
Young's modulus Y direction	9450 MPa
Young's modulus Z direction	9450 MPa
Poisson's Ratio XY	0.27
Poisson's Ratio YZ	0.4
Poisson's Ratio XZ	0.27
Shear modulus XY	5500 MPa
Shear modulus YZ	3900 MPa
Shear modulus XZ	5500 MPa
Tensile X direction	1979 MPa
Tensile Y direction	26 MPa
Tensile Z direction	26 MPa
Compressive X direction	−893 MPa
Compressive Y direction	−139 MPa
Compressive Z direction	−139 MPa
Shear XY	100 MPa
Shear YZ	50 MPa
Shear XZ	100 MPa

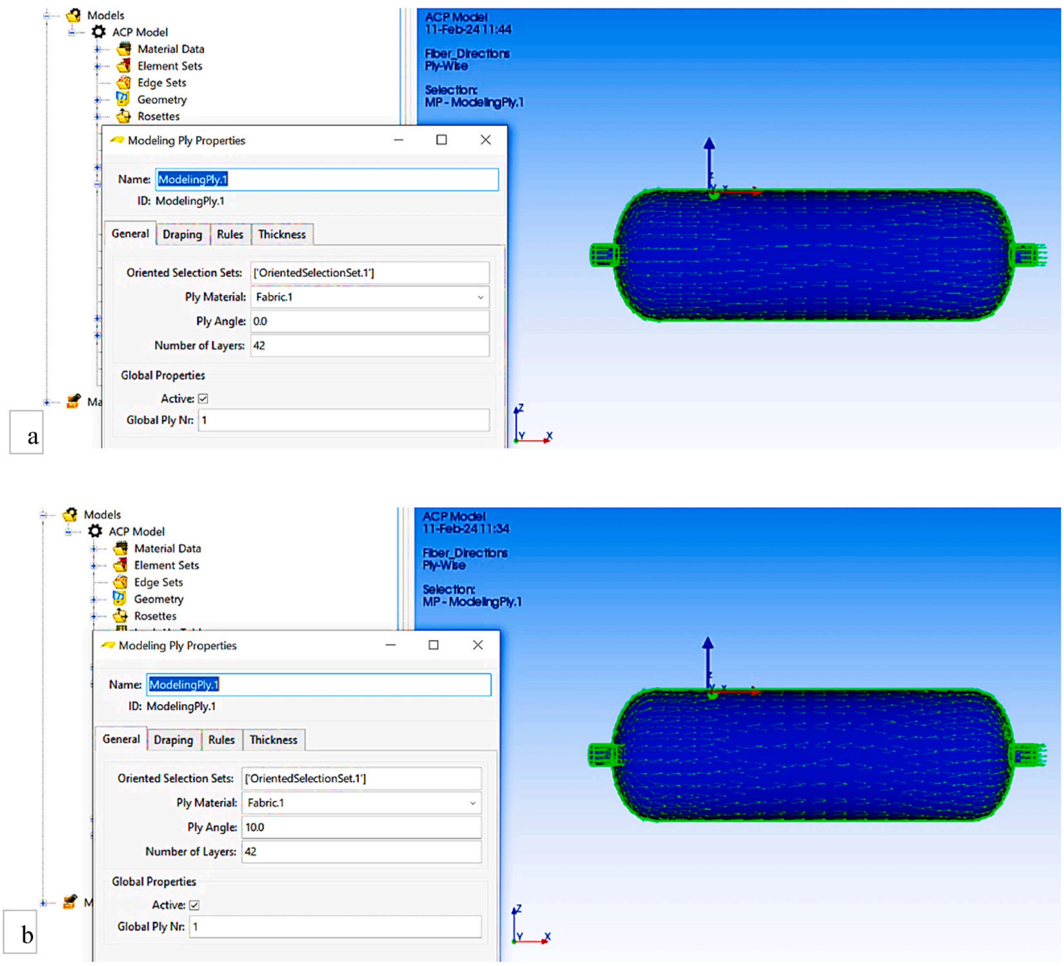

Fig. 7. Winding angles orientation on Ansys ACP Pre/Post module: (a) 0°, (b) 10°, (c) 80°, and (d) 90°.

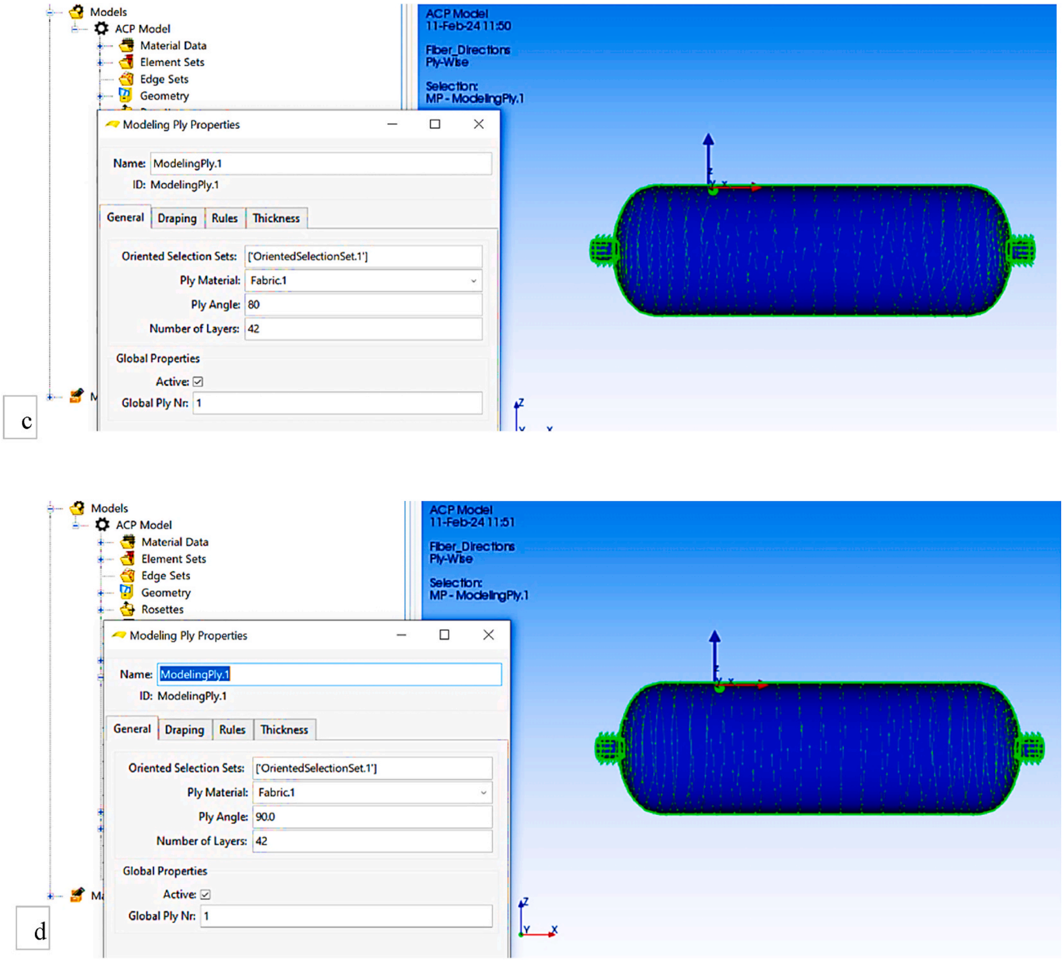

Fig. 7. (*continued*).

port. On the other port, there were free displacement conditions along the Z-directions and constant displacement conditions along the X- and Y-directions, as shown in Fig. 8 (a).

In FEA, mesh size has a critical role in determining the accuracy of the model. To achieve high-precision results and save time, an appropriate mesh size and type after many trials were selected. Quadrilateral dominant is a meshing method used to generate meshes where hexahedral elements dominate of 50 mm size (Fig. 8 (b)). Skewness and orthogonal quality are the two important metrics used in Ansys to assess mesh quality. To provide good mesh quality, low skewness in the 0 to 0.9 range and high orthogonal quality near or

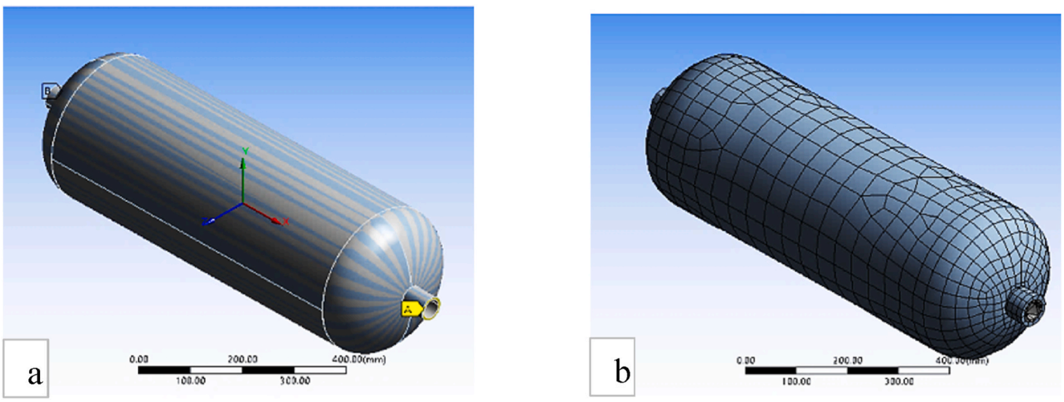

Fig. 8. Boundary conditions (a) and meshing (b) of FEM.

more than 0.1 are required. As illustrated in Fig. 9 a and b, respectively, the maximum skewness is about 0.4 and the orthogonal quality is approximately 0.98, indicating high mesh quality as per Ansys workbench criteria.

It was assumed that the nylon liner and composite wound material are bonded together to prevent sliding or separation during applying the internal pressure [17]. The internal pressure was applied uniformly up to 70 MPa, and the influence of the different winding configurations on the CPV's behavior was estimated versus stress and pressure.

3.3. Winding configurations

The effect of the individual winding angles at one orientation at different numbers of layers was studied in the range of 0°–90° on the maximum principal stress distribution and burst pressure (Table 3). Moreover, the influence of twelve winding patterns with combined winding angles indexed from A to L, as presented in Table 4, was also studied. A schematic representation of the fiber stacking arrangement for a winding angle of 10° is shown in Fig. 10 (a), and a winding pattern H with combined winding angles (see Table 4) is shown in Fig. 10 (b). For all angles in the current study, the layer's thickness is taken to be 0.5 mm.

3.4. Failure theory and burst pressure

The various failure theories for predicting the failure of materials, including the maximum principal stress theory and the equivalent stress theory, are used. The choice between these theories depends on whether the material is brittle or ductile. The maximum principal stress theory, which is frequently used in brittle materials, postulates that the failure of the material occurs when its uniaxial tensile strength is surpassed by the maximum principal stress. In other words, failure is specified by a fracture. Equivalent stress theory, also known as the von Mises criterion, is typically used for ductile materials. It suggests that failure occurs when the distortion strain energy (associated with deformations that change the shape but not the volume of the material) reaches a critical value. This theory is based on the idea that ductile materials do not fail due to states of stress that only impose a volume change. Therefore, the distortion strain energy is hypothesized to govern failure. As the studied Type 4 CPV is made of materials that behave in a brittle manner, the maximum principal stress theory is used in the current work to predict the value of burst pressure using Ansys Workbench's static structural module. According to this theory, the burst pressure of the CPVs is the pressure that corresponds to the ultimate tensile strength of the carbon-epoxy composite, which is provided in Table 2.

3.5. Netting analysis and Identifying the number of layers

The first part of the current study involves studying the effect of the single winding angles in one orientation to identify the angles that significantly influence the mechanical performance of the Type 4 CPVs. In this part, the number of layers gradually increased until it reached a value of maximum principal stress lower than the UTS of the composite materials. The minimum number of layers at which the CPV safely withstands the operating pressure was determined through trails using the Ansys ACP and static modules.

In the second part of this study, different proposed winding patterns utilizing combined angles were studied. To determine the minimum thickness and number of layers required to build a safe CPV consisting of both hoop and helical layers, the above-described method along with netting analysis methods were used. According to netting analysis, the CPV's stiffness and strength are imparted from the fibers. Because the matrix is not as stiff as reinforcement, its load-bearing contribution was ignored. This presumption is an excellent base for calculating composite thickness quickly, in addition to being conservative. The approximate layer thickness from netting analysis can be computed using the ultimate tensile strength of the epoxy-carbon unidirectional prepreg composite with the

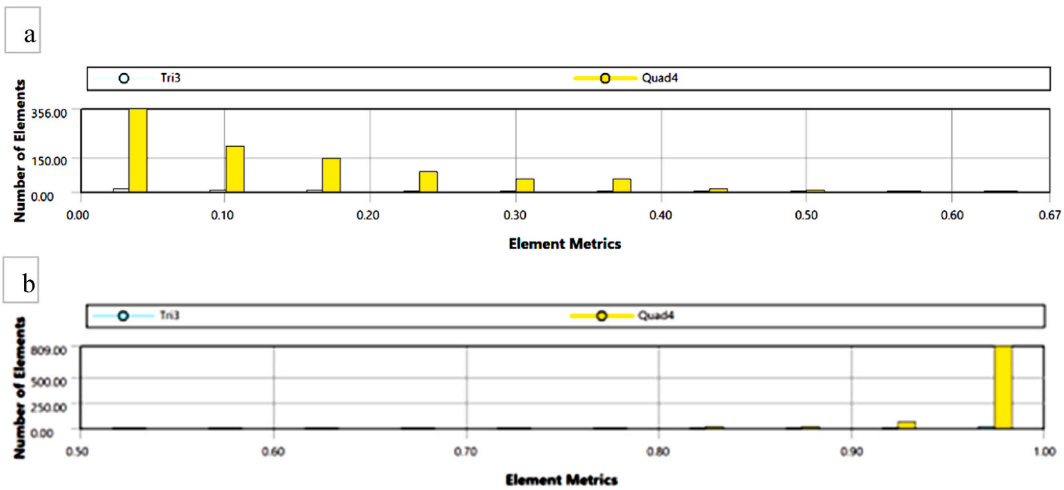

Fig. 9. Mech quality evaluation using skewness (a) and orthogonal quality (b) mesh metrics spectrums.

Table 3
A minimum number of layers, maximum principal stress, and burst pressure for the different winding angles.

Winding angle	Minimum number of layers	Max. principal stress (MPa)	The relation between the max. principal stress value (y) and the internal pressure (x)	Burst pressure (MPa)
0°	22	1944.7	y = 27.782x + 0.0008	71.23
10°	22	1898.9	y = 27.128x + 0.0025	72.95
20°	22	1728.1	y = 24.687x - 4E-13	80.16
30°	22	1581.1	y = 22.587x - 4E-13	87.62
40°	24	1720.1	y = 24.573x - 0.0058	80.55
50°	28	1868.7	y = 26.696x - 0.0042	74.13
60°	34	1955.9	y = 27.941x - 0.005	70.83
70°	40	1905.5	y = 27.222x + 0.01	72.70
80°	42	1882.6	y = 26.894x + 0.0017	73.59
90°	42	1874.2	y = 26.774x + 0.0083	73.91

Table 4
Winding patterns details.

Winding patterns index	Number of helical layers	Number of hoop layers	Total minimum number of layers	Winding pattern
A	8	13	21	$[90_3/10/\text{-}10/90_3/10/\text{-}10/90_3/10/\text{-}10/90_3/10/\text{-}10/90]$
B	10	13	23	$[90_2/20/\text{-}20/90_2/20/\text{-}20/90_2/20/\text{-}20/90_2/20/\text{-}20/90_2/20/\text{-}20/90_3]$
C	10	12	22	$[90_2/30/\text{-}30/90_2/30/\text{-}30/90_2/30/\text{-}30/90_2/30/\text{-}30/90_2/30/\text{-}30/90_2]$
D	10	8	18	$[40/\text{-}40/90_2/40/\text{-}40/90_2/40/\text{-}40/90_2/40/\text{-}40/90_2/40/\text{-}40]$
E	14	4	18	[50/-50/90/50/-50/50/-50/90/50/-50/50/-50/90/50/-50/90/50/-50]
F	20	–	20	[70/-70/70/-70/10/-10/70/-70/70/-70/10/-10/70/-70/70/-70/10/-10/ 70/-70]
G	26	–	26	[80/-80/20/-20/80/-80/20/-20/80/-80/20/-20/80/-80/20/-20/80/-80/ 20/-20/80/-80/20/-20/80/-80]
H	26	–	26	[60/-60/30/-30/60/-60/30/-30/60/-60/30/-30/60/-60/30/-30/60/-60/ 30/-30/60/-60/30/-30/60/-60]
I	36	–	36	$[60]_2[30]_2$ $[60]_2[30]_2$ $[60]_2[30]_2[60]_2[30]_2[60]_2[30]_2[60]_2[30]_2$ $[60]_2[30]_2$ $[60]_2[30]_2[60]_2[30]_2$
J	23	–	23	[40/-40/50/-50/40/-40/50/-50/40/-40/50/-50/40/-40/50/-50/40/-40/ 50/-50/40/-40/50]
K	19	–	19	[10/-10/20/-20/30/-30/40/-40/50/-50/60/-60/70/-70/80/-80/90/90/ 90]
L	26	–	26	[70/-70/20/-20/70/-70/20/-20/70/-70/20/-20/70/-70/20/-20/70/-70/ 20/-20/70/-70/20/-20/70/-70]

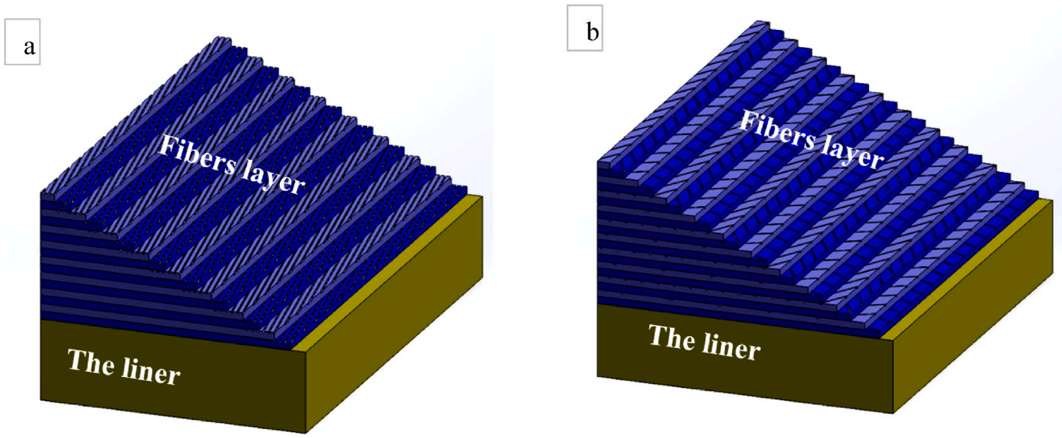

Fig. 10. Schematic representation of fiber stacking arrangement for (a) winding angle 10° and (b) winding pattern H.

following formulas [2,7,8,12,14,17]:

$$t_{helical} = \frac{PR}{2\sigma \cos^2 \theta} \quad \text{Eqn. (1)}$$

$$t_{hoop} = \frac{PR}{2\sigma}\left(2 - \tan^2\theta\right) \quad \text{Eqn. (2)}$$

Where t is the thickness of the wound composite, θ is the winding angle, P is the nominal working pressure (70 MPa), R is the radius of the vessel, and σ is the ultimate strength of the epoxy-carbon unidirectional prepreg composite (1979 MPa, see Table 2). The sum of the helical and loop thicknesses determines the layer's overall thickness.

3.6. Sensitivity analysis

Sensitivity analysis was used to determine how the pressure vessel's performance in terms of maximum principal stress is impacted by the winding angle and layer thickness. One of the simplest and most popular approaches to tracking changes in the output in sensitivity analysis is the one-factor-at-a-time (OAT) technique, which involves changing certain variables while keeping others unchanged. By making one adjustment at a time, all other parameters can be kept at their baseline or central levels. This enhances the comparability of the outcomes. Given that the one modified parameter will cause any changes observed in the output, this seems like a reasonable course of action. Furthermore, the variable that had the greatest influence can be identified [19].

In the current study, plotting the effects of the increase in winding angles at a constant number of layers and the increase in the thickness of layers at a constant winding angle on the value of maximum principal stress was conducted. The sensitivity indices were estimated by computing each relationship's slope. Because the relations are highly nonlinear, the authors utilize polynomial lines to calculate the sensitivity indices, maintaining the R-squared value above 0.9 to ensure the best possible correlation coefficient and regression line fit.

4. Results and discussion

4.1. The impact of the individual winding angle at different numbers of layers

The effect of each winding angle at one orientation, ranging from 0° to 90° at a rate of 10° increments every step, and the number of layers on the maximum principal stress were studied, and the results are presented in Fig. 11. The values of the stresses drop as the number of layers in the examined range, up to 42 layers, increases at all winding angles. This is explained by the fact that adding more composite layers makes the pressure vessel more durable, which reduces the generated residual stresses. These findings confirm previous results by Azeem et al. [1] and Di et al. [14].

At a constant number of layers, it can be observed that the maximum principal stresses are low at winding angles of 0°, 10°, 20°, and 30°, while at winding angles of 70°, 80°, and 90°, the maximum principal stresses are found to be high, and the intermediate stress levels are found at winding angles of 40°, 50°, and 60°. As a result, low helical winding angles (10°–30°) have a greater effect on enhancing the durability of the vessel than intermediate and high winding angles.

Theoretically, the ultimate tensile strength (UTS) of the material must be greater than the value of the maximum principal stress for the pressure vessel to be safe at the applied operating pressure. According to Table 2, the carbon fiber composite's UTS for the material under study is 1979 MPa. Several trials of increasing the number of layers and monitoring their impact on the generated maximum principal stress values were conducted using the Ansys ACP and static modules until a maximum principal stress value was lower than the UTS of the composite under study. Consequently, the minimum number of layers needed to guarantee a safe and lightweight CPV has been attained under operating pressure, as presented in Table 3. Fig. 12(a–j) shows the maximum principal distribution at the

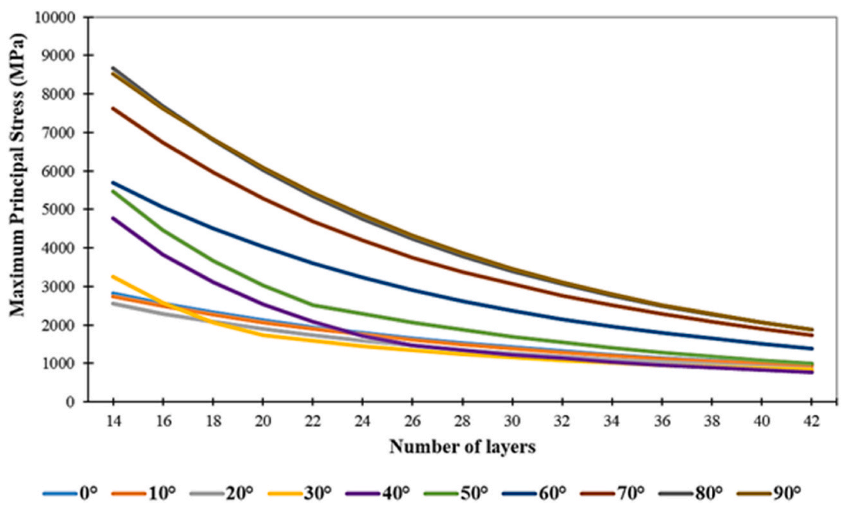

Fig. 11. Effect of the number of layers on the maximum principal stress distribution at different individual winding angles.

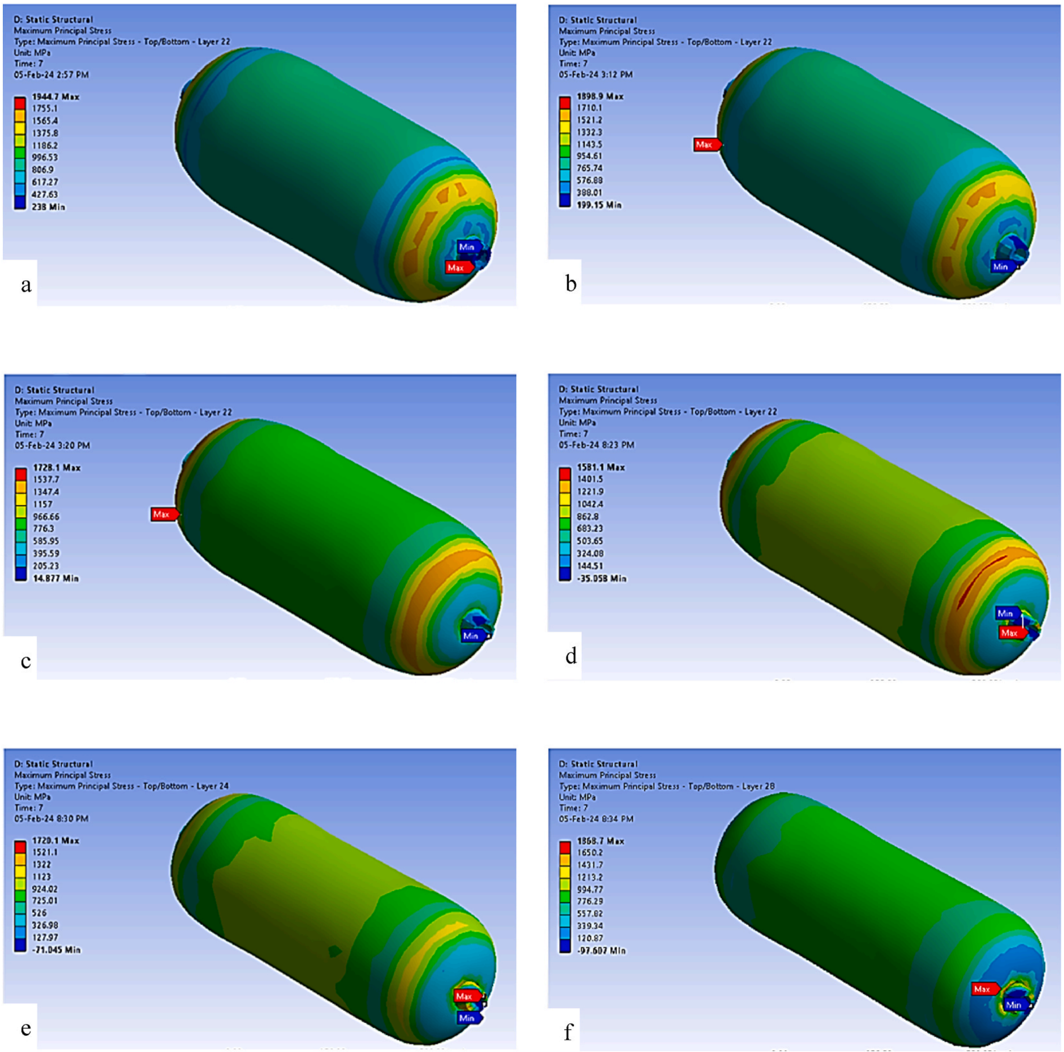

Fig. 12. Maximum principal stress distribution at constant working pressure and different winding angles (0°–90°) utilizing the minimum number of layers at the working pressure: (a) winding angle 0° - 22 layers, (b) winding angle 10° - 22 layers, (c) winding angle 20° - 22 layers, (d) winding angle 30° - 22 layers, (e) winding angle 40° - 24 layers, (f) winding angle 50° - 28 layers, (g) winding angle 60° - 34 layers, (h) winding angle 70° - 40 layers, (i) winding angle 80° - 42 layers, and (j) winding angle 90° - 42 layers.

minimum number of layers for the various winding angles at the applied working pressure of 70 MPa. These results may help the designers of CPVs in minimizing material waste by optimizing the weight efficiency of the vessel.

To make sure that the burst pressure is suitably higher than the working pressure of the compressed hydrogen gas, it is crucial to estimate the burst pressure at the attained minimum number of layers. Fig. 13 demonstrates the impact of varying internal pressure values up to 70 MPa on the stress the vessel experiences at various winding angles (0°: 90°). The linear equations that represent the mathematical relationship between the internal pressure and the maximum principal stress produced in the vessel at each winding angle are displayed in Table 3. These formulae were used to determine the value of the burst pressure, which corresponds to the stress value at the UTS of the examined composite.

As revealed in Table 3, all winding angles at the determined number of layers are relatively safe to operate at 70 MPa since their burst pressure falls between 70.83 and 87.62 MPa. It can be observed that the maximum principal stress is inversely proportional to the burst pressure of CPVs. This observation was previously reported [14,18]. The winding angle of 30° has the highest burst pressure of 87.62 MPa. Followed by 40° and 20° winding angles. These angles necessitate 22–24 minimum layers to be safe at the internal pressure of 75 MPa. Winding angles of 0° and 10° reveal lower burst pressure at the same number of layers. In contrast, angles that range from 50° to 90° need a greater number of minimum layers to be safe under the operating pressure, between 28 and 42, which leads to a higher-weight pressure vessel.

Therefore, the number of layers and the winding angles of the fibers are important design factors that affect many facets of the

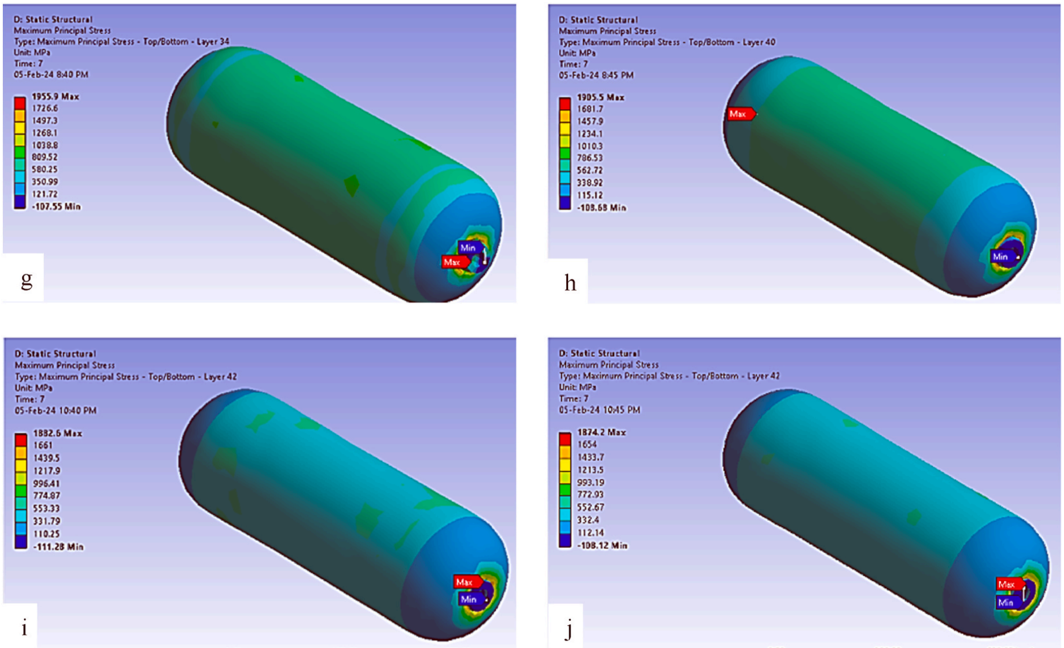

Fig. 12. (*continued*).

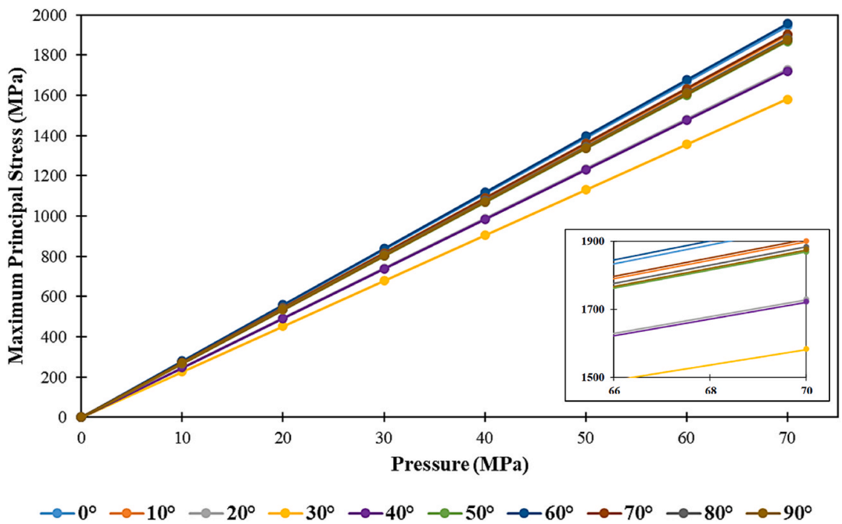

Fig. 13. Effect of the internal pressures on the maximum principal stress produced on the vessel at different individual winding angles (0°–90°).

mechanical performance of the CPVs. Increasing the number of composite layers improves the CPV's stiffness and strength and enhances load-bearing capacity. The number of layers also affects the burst pressure since more layers offer more reinforcement using a single winding angle. However, adding layers to a vessel wall can make it thicker and possibly heavier, which is important in applications where weight is an important consideration. Based on these findings, it can be concluded that low helical angles between 20° and 40° are useful for promoting the pressure vessel's durability and portability. When it comes to producing portable pressure vessels for transportation systems, these two aspects are quite important. It is worth mentioning that the stress distribution (Fig. 12) reveals that, using a single winding angle, the dome regions experience the highest stress values. As previously mentioned, the cylindrical portion of CPVs should be the intended point of failure to have a safe burst mode [4,9]. As a result, the following section examines the effects of several proposed winding patterns employing combined angles.

4.2. The impact of winding patterns with combined winding angles

The impact of twelve proposed winding patterns (Table 4) on the stress distribution and burst pressure was studied. To determine the minimum number of the hoop and helical layers—the value at which the maximum principal stress is less than the UTS of the composite under study—netting analysis was performed in conjunction with multiple Ansys static test trails.

The data provided in Table 4 shows that five winding patterns, A through E, combine low helical angles (10°–50°) with a hoop angle of 90°. The seven rest winding patterns, ranging from F to L, exclusively employ a combination of low and high helical angles. For the winding pattern from A to F, J, and K to be safe at the working pressure, a minimum number of layers, between 18 and 23, is needed. In contrast, 26 layers are needed for the winding patterns G, H, and L to be safe at the operating pressure. The winding pattern I requires the maximum number of layers (36) to ensure safety at the operating pressure.

These findings suggest that using a hoop angle in the winding pattern from A to E helps to lower the number of layers needed for the CPVs to be safe, which in turn affects the weight of the CPVs. The winding patterns D and E require the lowest minimum number of layers (18) to be safe at the operating pressure. Nevertheless, even though winding patterns H and I employ the same angles, 30° and 60°, the winding pattern I require more layers to be secure at operating pressure. The use of positive and negative winding angles is responsible for this. As a result, using both positive and negative winding angles has a favorable impact on the number of layers needed. The positive and negative helical winding angles, along with the hoop angle, are used in the winding pattern K, which is a combination of all winding angles from 10° to 90°. For this suggested winding pattern to be safe at a working pressure of 70 MPa, a low number of layers (19) is needed.

Table 5 presents the values of the maximum principal stress at the different winding patterns. Fig. 14(a–f) shows the distribution of the maximum principal stress in some selected winding patterns, namely winding patterns A, D, F, G, I, and L. As can be seen, the cylindrical section contains the crucial points of maximum stress; hence, a combination of winding angles provides a safe burst mode in contrast to using a single winding angle. Therefore, there is no metallic boss ejection during the explosion; instead, the entire cylindrical portion breaks along its circumferential layers [3].

Table 5 includes linear equations that show the mathematical relationship between the internal pressure and the maximum principal stress produced in the suggested winding patterns. These equations were used to determine the burst pressure value, which is equivalent to the stress value at the UTS of the composite that is being studied. The impact of varying internal pressure levels up to 70 MPa on the maximum principal stress that the vessel experiences at the various winding patterns under investigation is depicted in Fig. 15.

At low burst pressures around 70.5 MPa, the winding patterns A, F, I, and K fail. The winding patterns B to E, H, J, and L provide the burst pressure in the range of 72.66–77.99 MPa. Despite having the same winding angles, winding pattern H produces a higher burst pressure than pattern I. This is because the winding pattern H employs both positive and negative helical winding angles. Therefore, the winding pattern with only positive angles, or in one direction, has the lowest burst pressure, as observed. The winding pattern G, which employs 20° and 80° helical winding angles in both positive and negative orientations, exhibits the highest burst pressure of 81.26 MPa. This arrangement neutralizes the complex stress generated in wound composite layers, including hoop, axial, and shear stresses. This may be the reason for the exceptional performance of combining the positive and negative angles, i.e., the same angle in opposite directions. Furthermore, from a practical standpoint, using too many single-direction angles results in delamination between layers because of shear stresses.

The range of burst pressure when using a combination of solely helical angles without a hoop angle is the same as when using the hoop angle (90°) in combination with the low helical angle from 10° to 50°, according to the analysis of the results shown in Tables 4 and 5. Furthermore, it is advisable to employ intermediate helical angles of 40° and 50° with the hoop angle (90°) rather than low helical angles between 10° and 30°. These intermediate helical angles, which provide relatively similar or higher burst pressure at fewer layers to be safe at the operating pressure of 70 MPa, have a major impact on the vessel's weight.

When applied alone, as in Table 3, or combined with other high helical winding angles of 70° (winding pattern L) and 80° (winding pattern G), or with the hoop angle (winding pattern B), as in Table 4, the winding angle of 20° demonstrates exceptional performance. Employing a 20° winding angle for the CPVs to be safe at the operating pressure reveals high burst pressure at a low number of

Table 5
A maximum principal stress and burst pressure for the different winding patterns.

Winding patterns index	Max. principal stress (MPa)	The relation between the max. principal stress value (y) and the internal pressure (x)	Burst pressure (MPa)
A	1972.9	$y = 28.185x - 0.0008$	70.21
B	1849.1	$y = 26.416x - 0.0042$	74.92
C	1867.4	$y = 26.678x + 0.0025$	74.18
D	1858.2	$y = 26.546x - 0.0033$	74.55
E	1776.1	$y = 25.372x - 0.0025$	77.99
F	1960.5	$y = 28.007x - 0.0025$	70.66
G	1704.8	$y = 24.354x$	81.26
H	1877.3	$y = 26.818x - 0.0058$	73.79
I	1960.1	$y = 28.002x + 0.0008$	70.68
J	1906.5	$y = 27.236x - 0.0033$	72.66
K	1957.9	$y = 27.97x$	70.75
L	1804.9	$y = 25.785x - 0.0008$	76.75

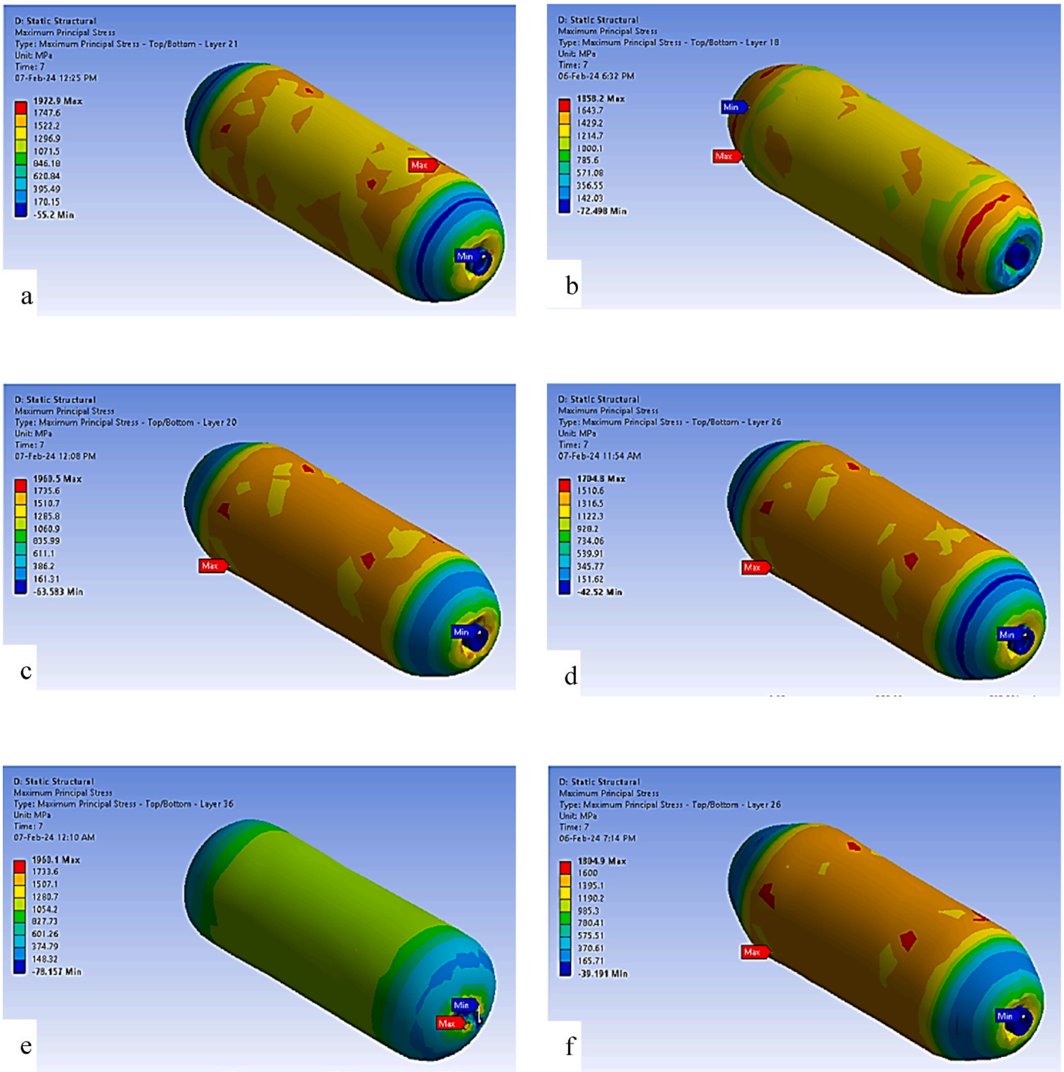

Fig. 14. Maximum principal stress distribution of some selected winding patterns: (a) winding pattern A, (b) winding pattern D, (c) winding pattern F, (d) winding pattern G, (e) winding pattern I, and (f) winding pattern L.

necessary layers. Following the winding angle of 20°, the winding angles of 30° and 40° have a good impact on performance as they exhibit low winding layer counts and high burst pressure.

Industrially, these winding angles are easily applied in the filament winding business because they do not cause slippage, unlike polar angles or extremely low helical winding angles, which are typically between 0 and 10°. Furthermore, in terms of practicality, the hoop winding angle (90°) can be accomplished on a single moving pass of the carriage, but the helical angles require multiple back-and-forth movements of the payout eye. As a result, applying the hoop angle reduces production time and boosts productivity.

4.3. Sensitivity analysis of winding angle and thickness of layers variation

Sensitivity analysis is a technique to identify the input factors that most influence an output behavior and to understand how the model output depends on the model input. One of the most popular methods for conducting sensitivity analysis is the one-factor-at-a-time (OAT) technique, which involves changing one variable at a time and observing the effect on the output. The so-called sensitivity indices of the input variables are computed and analyzed in relation to a specific quantity of interest in the model output. These indices give an indication of how much changes in a particular parameter affect the result. As a consequence, the sensitivity index illustrates a parameter's relative importance. Greater sensitivity indices signify a substantial impact of a parameter on the result, while lower indices imply a lesser influence. The rate at which the model's output changes in relation to a certain parameter is referred to as the slope. Greater sensitivity is implied by a steeper slope, and lower sensitivity is implied by a flatter slope [19].

An OAT sensitivity analysis of the impact of variation in the winding angle on the maximum principal stress at a constant number of

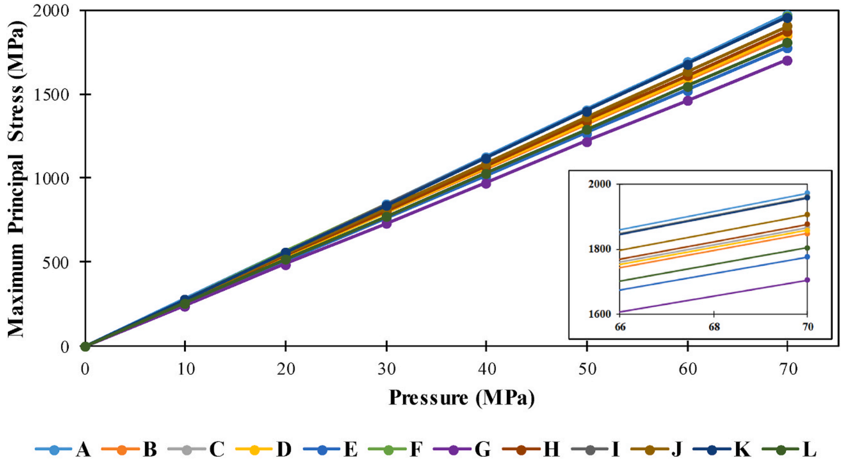

Fig. 15. The maximum principal stress produced on the vessel at different values of the internal pressures utilizing the suggested winding patterns.

layers using the linear regression method is shown in Fig. 16. The analysis was performed at a constant number of layers at 22, 34, and 42. As the relation between the winding angle and maximum principal stress is highly non-linear, the polynomial lines were utilized to calculate the sensitivity indices, keeping a high R-squared value to ensure the best possible correlation coefficient and regression line fit. A comparable response was observed at different numbers of layers. From a 0–30° winding angle, the increase in the winding angle causes a decrease in the maximum principal stress with a relatively weak sensitivity index in the range of 4.45–12.61 MPa/deg. Therefore, an enhancement in the pressure vessel performance was brought about by the increase in the burst pressure. As previously indicated, the maximum principal stress is inversely proportional to the burst pressure. Increasing the winding angles in the range of 30–80° results in an increase in the maximum principal stress with a high sensitivity index in the range of 24.52–79.11 MPa/deg.

A weak response was seen from an 80–90° winding angle, with a sensitivity index between 0.9 and 9.42 MPa/deg. As observed, increasing the number of layers reduces the influence of the winding angles on the maximum principal stress, as noted by the sensitivity index.

Fig. 17 presents the OAT sensitivity analysis of the impact of variation in the thickness of layers on the maximum principal stress using the linear regression method. The analysis was performed at a constant winding angle of 20, 60, and 90°. Once again, the relation between the thickness of layers and maximum principal stress is highly non-linear; the polynomial lines were used to calculate the sensitivity indices. In this instance, the maximum principal stress constantly falls as the thickness of the layers increases. At the various

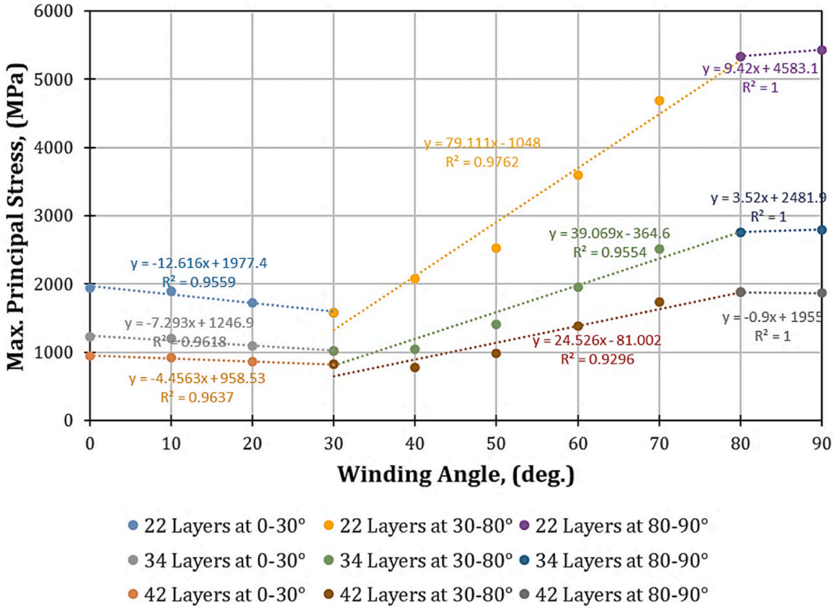

Fig. 16. OAT sensitivity analysis of the winding angle: The impact of variation of the winding angle at a constant number of layers on the maximum principal stress using the linear regression method.

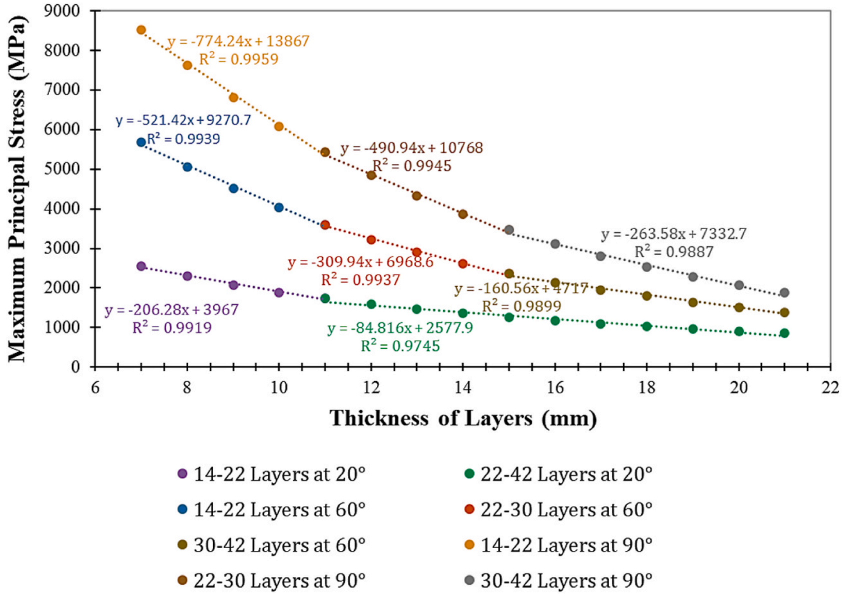

Fig. 17. OAT sensitivity analysis of the thickness of layers: The impact of variation in the thickness of layers at a constant winding angle on the maximum principal stress using the linear regression method.

winding angles, a similar response was noted. As can be seen from the sensitivity index, lowering the winding angles diminishes the impact that the thickness of the layers on the maximum principal stress.

Comparing the sensitivity indices in Figs. 16 and 17, it is found that the maximum principal stress is more strongly influenced by the thickness of layers than by winding angle, as indicated by a higher sensitivity index (84.71–774.24 MPa/deg).

5. Conclusions

The impact of winding angles on the filament wound Type 4 CPVs' stress distribution and burst pressure predictions were investigated using the Ansys ACP Prep/Post and static modules. An analysis has been performed on the effects of both individual and combined winding angles. Finding the angles that have the most significant impact on weight and burst pressure is the aim of this work. The mobility and performance of the CPVs can be greatly enhanced by weight reduction, which also makes the use of hydrogen fuel in transportation systems more feasible. Fuel can be securely stored as a compressed gas at a working pressure of up to 70 MPa using the proposed Type 4 CPVs. By modifying the fiber winding angles, arrangement, and number of layers, Type 4 CPV can be made to fit specific service circumstances while guaranteeing reliable performance. This paper offers an analytical framework for the cost-effective design and utilization of high-pressure Type 4 CPVs in hydrogen storage fields. The following conclusions are drawn.

- By tailoring the number of layers and the orientation and value of the fibers' winding angle, the stress distribution in the CPVs and the intended point of failure to have a safe burst mode can be controlled.
- The weight and burst pressure of the CPVs are greatly impacted by the orientation of the fiber angles since the combination of positive and negative angles greatly promotes burst pressure at lower weights, enhancing the mobility of the vessel and reducing the material cost.
- High-efficiency CPVs can be achieved by employing hoop and intermediate helical angles (40° and 50°). Combining high and low winding helical angles yields comparable results.
- The performance of CPVs was found to be strongly influenced by the thickness layers rather than by winding angle, as evidenced by a higher sensitivity index. As the number of layers increases, the influence of the winding angle on the maximum principal stress decreases, and as the winding angle decreases, the influence of the thickness of layers on the maximum principal stress decreases, as indicated by the OAT sensitivity analysis.

Ethical approval

This article does not contain any studies with human participants or animals performed by any of the authors.

Data availability statement

Data will be available on request.

CRediT authorship contribution statement

Reham Reda: Writing – review & editing, Project administration, Conceptualization. **Mennatullah Khamis:** Writing – original draft, Software. **Adham E. Ragab:** Supervision, Resources, Funding acquisition. **Ahmed Elsayed:** Resources, Conceptualization. **A.M. Negm:** Software, Conceptualization.

Declaration of competing interest

No potential conflict of interest was reported by the author(s).

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